ENGINEERING CASE LIBRARY

ECL 1003

Impulse Turbine Blade Failures

Abstract

The catastrophic failure of impulse blades in large turbines in the 1930's necessitated a good deal of analytical and experimental work before the nature of the problem was well understood. Reducing the problem to its essentials, the case study deals primarily with the analysis of turbine blade vibration due to intermittent steam loading. The case history deals also with some aspects of failure diagnosis, with fatigue under combined stress, with some experimental techniques, and with design improvements that have resulted from the analytical and experimental findings.

The material is presented in such a fashion that the student can work out several relevant problems for himself.

Prepared with the support of the National Science Foundation. Written by R. P. Kroon, Professor of Mechanical Engineering, University of Pennsylvania. The co-operation of the Westinghouse Electric Corporation and particularly that of Mr. W. E. Trumpler, Manager, Mechanical Design Section, Large Turbine Division, in making available the engineering information included herein, is gratefully acknowledged. Many engineers and mechanics worked together on the project reported here. Dr. C. C. Davenport, now Director of Reliability, Rocketdyne Division, North American Aviation, Inc., and Mr. O. E. Rodgers, now Manager of Engineering Research, Scott Paper Co., were two of the principal collaborators of the writer.

General Background

In the 1930's the introduction of steam turbines, operating at higher pressures, higher temperatures and higher r.p.m. than those, on which several decades of service had been experienced, gave rise to some catastrophic failures of the turbine blades. These failures, and the analytical and experimental program which was carried out to learn how to design safe blades cost the industry many millions of dollars. This case study is concerned with:

(a) failure diagnosis

(b) vibration analysis

(c) experimental observations

(d) design improvements

History and General Design

Particularly since the turn of the century, the turbo-generator has played an ever increasing role in providing the nation with electric power. For many years, the power consumption in the United States has doubled about every 10 years. By the 1930's, metallurgical developments and mechanical improvements made it possible to go to high steam temperatures and pressures (typically 900° F, 1250 psi at turbine inlet). Because there were many successful turbines in operation at lower inlet conditions (say 650°F, 600 psi), the idea was conceived to install "superposed turbines", employing the higher available inlet conditions and exhausting into the existing steam turbines. This history pertains primarily to these superposed turbines, but the blade problem discussed here applies to many modern large steam turbines.

The impluse blading of a modern steam turbine is shown in Figure 1. Steam enters by way of a number of control valves through a pipe, feeding into a number of nozzle passages. Each control valve thus admits steam only over a fraction of the circumference. This feature, known as "partial admission", makes it possible to increase the area of admission as the load demand goes up. It results in better part-load efficiency than if the steam were admitted over the entire periphery at all loads. The blading in the control stage in the turbines is of the type known as a Curtis stage (Figure 2). There are two rotating blade rows, in between which there is a stationary element.

The first rotating blades installed were of the "straddle T" design shown in Figure 2 and were 1 1/2 inch wide. They lasted only 17 hours. Figure 3 presents a view of the failure. It can be seen that several blades broke off at the top of the root; one of the blades shown (and many others) failed at the lower fillet of the root. The designer, under pressure to get the turbines back in service, replaced the 1 1/2 inch blades with much stronger "double T" type blades, 2 inches wide, shown in Figure 4. These blades lasted only 23 hours.

Lesson: When a design fails, "beefing it up" is not always the answer. In important cases, a thorough analysis is in order.

The severity of the situation was now realized and an extensive program was launched to find out the reason for the failures and to produce safe designs for the future.

Questions: What type of failure is this? Is it due to excessive steady stress? Is it due to creep of the metal at high temperature? Is it due to vibration?...

Considerations: At the point of failure (at the fillet in the T of the root) the nominal design stresses were:

	1 1/2 inch	2 inch
direct stress due to centrifugal force	10,000 psi	4,700 psi
bending stress due to steam load	16,200 psi	3,090 psi

The material, 12% chrome steel, has the following properties at temperatures to which the blade root is exposed:

fatigue limit	43,000 psi
yield stress	70,000 psi
uniaxial creep stress causing rupture in 10^3 hours	46,000 psi
uniaxial creep stress causing rupture in 10^5 hours	33,000 psi

The Goodman Diagram

One approach to evaluate the design safety under combined stress is that of the Goodman Diagram (Figure 5a). Here the steady stress which produces failure (yielding, or creep rupture) is plotted along the abscissa, and the

and the endurance limit (the stress which the material can withstand when subjected to alternating load) along the ordinate. When direct stress and alternating stress are combined, it is presumed that a safe design would be represented by any point lying within a limit curve, while any point outside the curve would signify failure. For most situations, the limit curve can be taken to be elliptical.

If we assume, that the blading is stressed only by centrifugal force and by the pulsating steam load, we would have, for the 1 1/2 inch blades, a steady stress of 10,000 psi, superposed on which would be a pulsating stress* of 16,200 psi. This would be equivalent to a nominal steady stress of $10,000 + \frac{16,200}{2} = 18,100$ psi, superposed on which would be a nominal alternating stress of $\frac{16,200}{2} = 8,100$ psi. (Where stress concentration exist, this nominal alternating stress must be multiplied by a stress concentration factor).

On the basis of such figures, it appeared to the designer that his design was quite safe.

Failure Diagnosis: The following considerations seemed significant:

(1) The failures showed no reduction of area at the cross section.

(2) Steady direct and bending stresses can be computed rather accurately. It did not seem that these stresses could by themselves cause failure in such a short time.

(3) Unless the rather sharp fillets produced a very large effect on the creep resistance, the very short time in which failure occurred ruled out failure due to creep per se.

(4) The fracture surfaces show the typical "beach mark" pattern of progressive fatigue.

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*The word "pulsating stress" is used for a stress varying from zero to maximum value (16,200 psi in this case) as contrasted to an "alternating stress" which varies from negative to positive.

Conclusion:

It was concluded that the failures were caused primarily by some sort of vibratory loading. The mystery was how vibratory stresses of a magnitude large enough to cause fâilure could occur, particularly on the stronger re-design.

Vibration Analysis

It was surmised, that in some manner the phenomenon of passing through the steam jet in partial admission was responsible for the high vibratory stresses.

The following facts were established:

(1) The lowest natural frequency of the 1 1/2 inch wide rotating blades was in the order of 1080 cycles per second. At 3600 rpm, or 60 rps, this corresponds to 18 cycles of vibration per revolution.

(2) The damping in the structure is quite small, with a "logarithmic decrement" (the natural logarithm of the ratio of successive amplitudes in free vibration) being in the order of 0.02.

For simplicity the vibration analyses were based on a force diagram shown in Figure 5. Also for simplicity, the system was taken to have only one degree of freedom. It was thought that the natural frequency of the blades would vary sufficiently that:

(a) there could be a whole number of cycles per revolution.

(b) the phase relations would be such that the instant of loading (t_1) and unloading (t_2) might occur in such a way as to produce the maximum amount of vibration.

Assignment: Derive an expression for the vibratory amplitudes of a system of mass m, spring constant k, exciting force F according to Figure 5, and n cycles per revolution, with the above assumptions (steady state conditions). See if you can simplify your answer by assuming that is a very small number. See Appendix A for one way to solve this problem.

Experimental Procedure

It appeared most necessary to confirm this theory by experimental measurement of the blade vibration. In addition, it was hoped that these

experiments would give some information on the actual loading diagram and the damping of the blade system under high centrifugal load.

The recording system had to operate in a centrifugal field of about 10,000 g and in a dry steam atmosphere. At the time of the failures, strain gauges operating at 800-900° F had not yet been developed.

An optical system (Figure 6) was developed, in which the angular displacement at the blade tip deflects a light beam focusing on a photographic plate. Without vibration, the image is circular. Blade vibration causes the light beam to deviate from the circle, resulting in a record of the type shown. The small mirror mounted at the tip of the blade is made from stellite, and ground with a slight curvature. To try out the optical method, a small experimental system (Figure 7) was built, in which the blade vibration was excited by passing through the field of a strong magnet. The system was found to work adequately and the blade vibration recorded was of the type expected. The very first experimental record made (with simulated steam jet) is reproduced in Figure 8. It gave a real thrill to the men who had designed and built the experimental apparatus.

A full scale experimental steam turbine was installed in the Schuylkill station of the Philadelphia Electric Company. The purpose of this co-operative test program was to measure the blade vibration under actual steam conditions with full size blades, using the optical system. The rotor of this experimental turbine is shown in Figure 9.

General Conclusion

Both analysis and experiment showed that severe vibration with amplitudes several times the "static deflection", corresponding to a steady load F, can occur when impulse blades operate with partial admission.

Question: Suppose that the vibration record is as shown in Figure 10a and that the force in the middle of the steam jet can be computed with satisfactory accuracy. Assume further that the damping forces are small compared with the inertia and spring forces in the system. Can you indicate a simple way to find points on the force-time diagram? Appendix B shows one way in which this can be done.

It was found experimentally, that the circumferential force diagram for a simple blade has a shape as shown in Figure 10b. The flow phenomena involved (non-steady compressible flow with velocities often greater than sonic velocity) are difficult to analyze quantitatively but a qualitative analysis is not difficult. The phases I-VI of Figure 10c relate to corresponding portions of the force diagram.

I. The blade moves through "stagnant" steam with negligible force exerted on it.

II. Steam enters the passage ahead of the blade, accelerating the stagnant steam and exerting additional pressure on the forward side of the blade. This results in a negative blade force.

III. Conditions in the passages on both sides of the blade are approaching steady conditions.

IV. Conditions are stabilized in the middle of the jet.

V. The passage to the rear of the blade receives full flow, the passage ahead is emptied by inertia effect so that the pressures are reduced. The net result is a higher than average force on the blade.

VI. The pressures ahead of the blade approach stagnant conditions, those in the passage behind the blade are less. The force on the blade is negative.

In cases with predominantly steady stresses, and ductile materials, stress concentrations may not be severe, as by means of a small amount of a yielding the stress field can adjust itself. But in fatigue, where the stresses are below the yield point, high stress concentrations can lead to disaster.

In the case of impulse blades, the situation is particularly severe, because just beside the small fillet in the root there is applied a high contact load (to hold the blade in the rotor).

Fatigue tests on actual blades held in a simulated rotor showed the surprisingly high stress concentrations of 3.5 (1 1/2" blade) and 2.0 (2" blade).

Lesson: Avoid sharp corners in highly stressed parts.

Design Improvements

From the information gained, it is obvious that besides reducing the steam loading there are a number of ways in which design improvements can be made. For example, it is clear from the analysis that in general the following will help:

(a) Increasing the natural frequency of the blade structure.

(Note that at the large number of cycles per blade it is not practical to try to "detune" the blades. Temperature variations, tolerances, variations in fit between rotor and blade, cause a fair amount of scatter.)

(b) Increasing the damping.

(c) Making the loading more gradual. One of the best ways to accomplish this is to connect several blades together by means of a "shroud band", riveted or welded to the blade at the tip. The number of blades that can be connected together is limited by thermal stresses. The reason that the much stronger 2" blades failed just about as quickly as 1 1/2" blades was that they were not shrouded. (The designer did not think he could afford the time to develop the technique to manufacture a riveted shroud on such large blades.) Another way is to design the nozzle passages to "cushion" the shock of entering and leaving the steam jet.

(d) Increasing the blade strength (reducing nominal stresses and reducing stress concentration).

Problem: See if you can design a better blade fastening.

Epilogue:

The Westinghouse Electric Corporation carried out a very extensive experimental program with various blade fastenings.

A multiple blade fastening, in which three blades were brazed together and held by pins at the edges (Figure 11c) showed extremely high frequencies and low stress concentrations. It was used in several new machines, but was

found unsatisfactory in the long run due to creep at higher temperatures.

A "side entry" design with shroud band is shown in Figure 12. This design permits a large section modulus and low nominal stress at the point of failure. It is used in many modern turbines.

A variation of the design illustrated in Figure 11c is shown in Figure 13. Here the individual blades are welded together. The number of pins, by which the blade block is held in the rotor, is here increased to three.

A material with greater inherent damping (NIVCO) was developed and is in use today.

The analysis presented here considers only the lowest circumferential mode of vibration, simulated by a system of one degree of freedom. The computations have been extended for several modes of vibration of blade groups, in circumferential and axial direction.

The demand for operation at higher temperatures and pressure has continued. Some large turbines now operate at 1000° F, 3500 psi inlet conditions.

Appendix A

A sudden increase or decrease in the force on the blade is equivalent to a shift of the neutral point about which the vibration takes place. Let x_{st} represent the steady deflection which a force F produces. It is easily seen that the largest vibration motion will take place, when F is applied at the instant when the blade is at its extreme negative position, and when F is removed when the blade is at its largest positive deflection. The motion of the blade, plotted against time, will then look as shown in Figure 14. Note that n_1 and n_2 are not integers.

Various amplitudes during the revolution are then related as follows:

$$b = a + x_{st}$$
$$c = b\, e^{-n_2\delta}$$
$$d = c + x_{st}$$
$$a = d\, e^{-n_1\delta}$$

These equations are sufficient for the unknown amplitudes a, b, c, d. We find for example:

$$b = x_{st}\,\frac{1 + e^{-n_1\delta}}{1 - e^{-n_1\delta}} \qquad d = x_{st}\,\frac{1 + e^{-n_2\delta}}{1 - e^{-n\delta}}$$

For fatigue we are primarily interested in the amplitude range which will be denoted as $2\,x_{osc} = b' + x_{st} + d'$.

Since $b' = b\, e^{-\delta/2}$ and
$d' = d\, e^{-\delta/2}$, we have

$$x_{osc} = x_{st}\;\frac{1}{2} + \frac{(2 + e^{-n_1\delta} + e^{-n_2\delta})e^{-\delta/2}}{2(1 - e^{-n\delta})}$$

$$x_{osc} = x_{st}\;\frac{1}{2} + \frac{2 + e^{-n_1\delta} + e^{-n_2\delta}}{2(1 - e^{-n\delta})e^{\delta/2}} \qquad (1)$$

The logarithmic decrement in this case is a very small quantity ($\delta \ll 1$), and its nature is such that it is difficult to ascertain its value with great accuracy. Therefore, it is logical to see if equation (1) cannot be simplified.

We will make use of the series: $e^y = 1 + y + \frac{y^2}{2} + \frac{y^3}{6} \ldots$

and for convenience we will write $n_1 = \gamma n$, with $0<\gamma<1$, so that $n_2 = (1-\gamma)n$.

Equation (1) can then be written:

$$x_{osc} = x_{st} \quad \frac{1}{2} + \frac{2 + \quad 1 - \gamma n\delta + \frac{(\gamma n)^2}{2} \delta^2 \ldots \quad + \quad 1 - (1-\gamma)n\delta + \frac{(1-\gamma)^2 n^2}{2} \delta^2 \ldots}{2 \quad n\delta - \frac{(n\delta)^2}{2} + \frac{(n\delta)^3}{6} \ldots \quad 1 + \frac{\delta}{2} + \frac{\delta^2}{8} \ldots}$$

Arranging the terms in orders of δ:

$$x_{osc} = x_{st} \quad \frac{1}{2} + \frac{4 - n\delta + (\gamma^2 n^2 - \gamma n^2 + \frac{n^2}{2}) \delta^2 \ldots}{2n\delta \quad 1 - \frac{n-1}{2}\delta + \frac{n^2}{6} - \frac{n}{4} + \frac{1}{8} \quad \delta^2 \ldots}$$

Carrying out the division:

$$x_{osc} = x_{st} \quad \frac{1}{2} + \frac{1}{2n\delta} \quad 4 + (n-2)\delta + (\gamma^2 n^2 - \gamma n^2 + \frac{n^2}{3} - \frac{n}{2} + \frac{1}{2}) \delta^2 \ldots$$

If we consider, that the number of cycles per revolution n may range perhaps from 15 to 100, we can furthermore neglect the terms $\frac{n}{2}$ and $\frac{1}{2}$ compared with those containing n^2.

This leads to the simplification:

$$x_{osc} \approx x_{st} \quad 1 + \frac{2}{n\delta} - \frac{1}{n} + (\frac{\gamma^2}{2} - \frac{\gamma}{2} + \frac{1}{6}) n\delta \ldots$$

Examining the relative values of the terms in this equation, we dare to go one step further and write:

$$x_{osc} \approx x_{st} \left(1 + \frac{2}{n\delta}\right) \qquad (2)$$

The approximative formula (2) is accurate to less than 1% for values of $n\delta$ less than 0.5. Since the decrement δ is almost never known with great accuracy, equation (2) is acceptable in most cases. Its simplicity permits us to draw some conclusions:

(a) The damping and the number of cycles per revolution are of prime importance. It is the product $n\delta$ that matters.

(b) The arc of the jet (expressed by γ) is not too significant.

The exact formula (1) shows that the oscillating stresses are somewhat higher for small arcs (always assuming that the timing of the shocks is the worst possible).

(c) It is particularly disadvantageous to have the blades go through two or more jets per revolution, as was done in some early designs. Having two jets is equivalent to cutting the quantity $n\delta$ in half.

(d) The change from 1800 rpm of the older turbines to 3600 rpm for the newer turbines was extremely important. For the same steam flow and the same blade velocity, the active portion (the "port" of the blade) is twice as high for the 3600 rpm blade. This means that the 3600 rpm blade has twice the steam force, a much lower natural frequency, while of course the time for one revolution is only 1/2 of that of the 1800 rpm. In a typical case the vibratory stresses in a 3600 rpm blade are 10 times those of the corresponding 1800 rpm blade!

Lesson from series approximation: Unless great accuracy is called for, a better physical grasp of a problem can often be obtained by neglecting quantities of higher order.

Appendix B

Neglecting damping, the differential equation of the assumed system with a single degree of freedom with a variable steam force f (t) is:

$$m\ddot{x} + kx = f(t) \quad \text{or} \quad \frac{m\ddot{x}}{k} + x = \frac{f(t)}{k} \quad .$$

At inflection points of zero curvature $\ddot{x} = 0$; thus at these points $x = \frac{f(t)}{k}$. (See Figure 15).

Therefore the points A, B, C, D, E, correspond to points on the force curve with the scale relation $f(t) = \frac{Fx}{x_{st}}$, where F is the steady steam load in the center of the steam jet, and x_{st} the steady deflection which would be produced by F.

Note: It is difficult to determine the points of inflection with good accuracy.

Problem: With modern instrumentation, acceleration can be conveniently measured electronically. Suppose that you could measure $\ddot{x}$. Can you think of an electronic network, that would have $\ddot{x}$ as input and $\frac{f(t)}{m}$ as output, assuming that the natural frequency of the system is known?

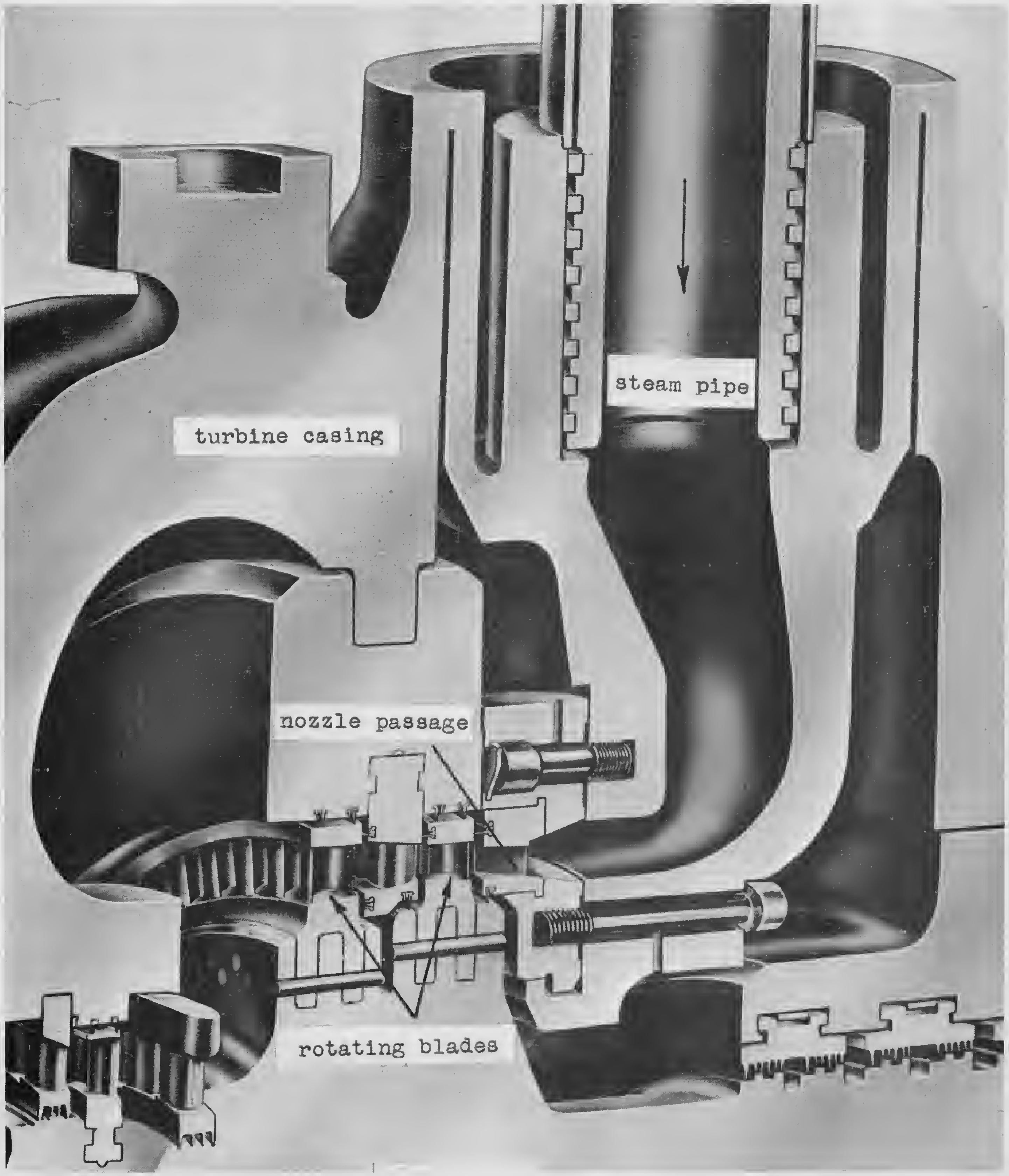


Figure 1
Cut open view of superposed turbine, showing impulse stage.

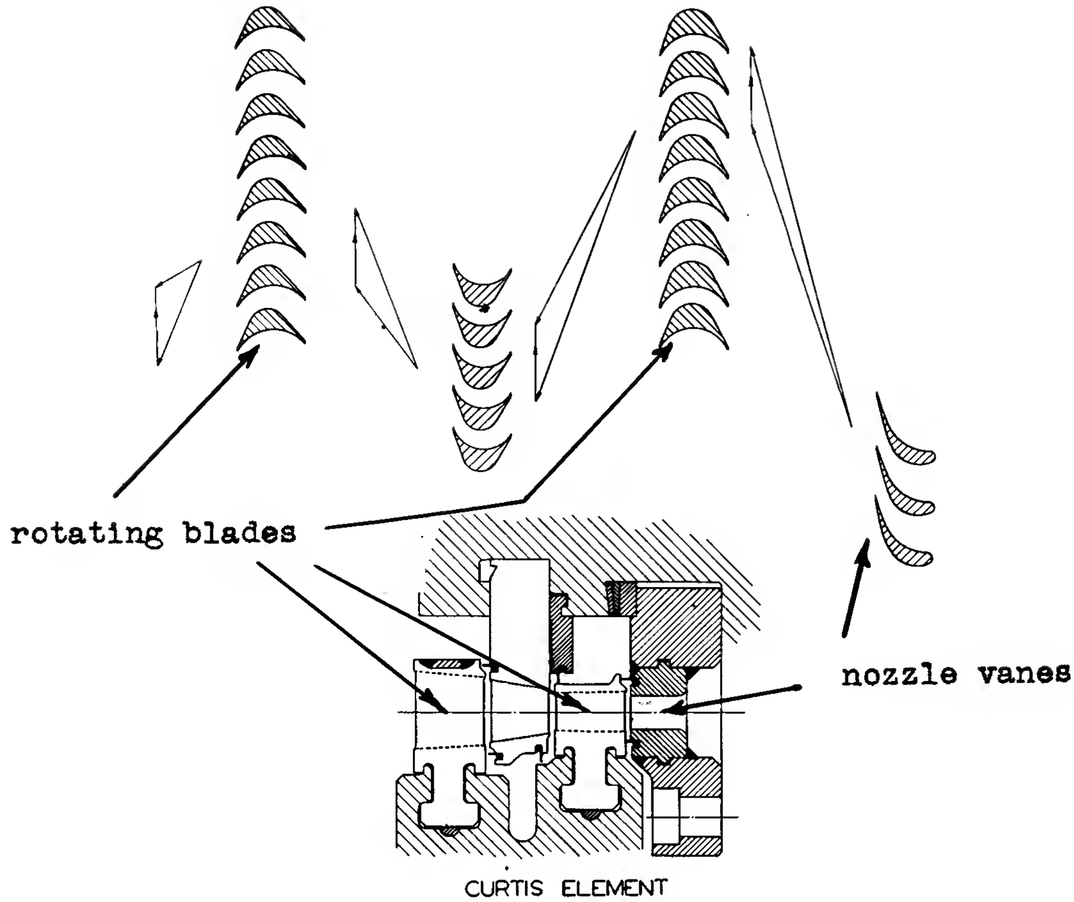


Figure 2 Curtis element.

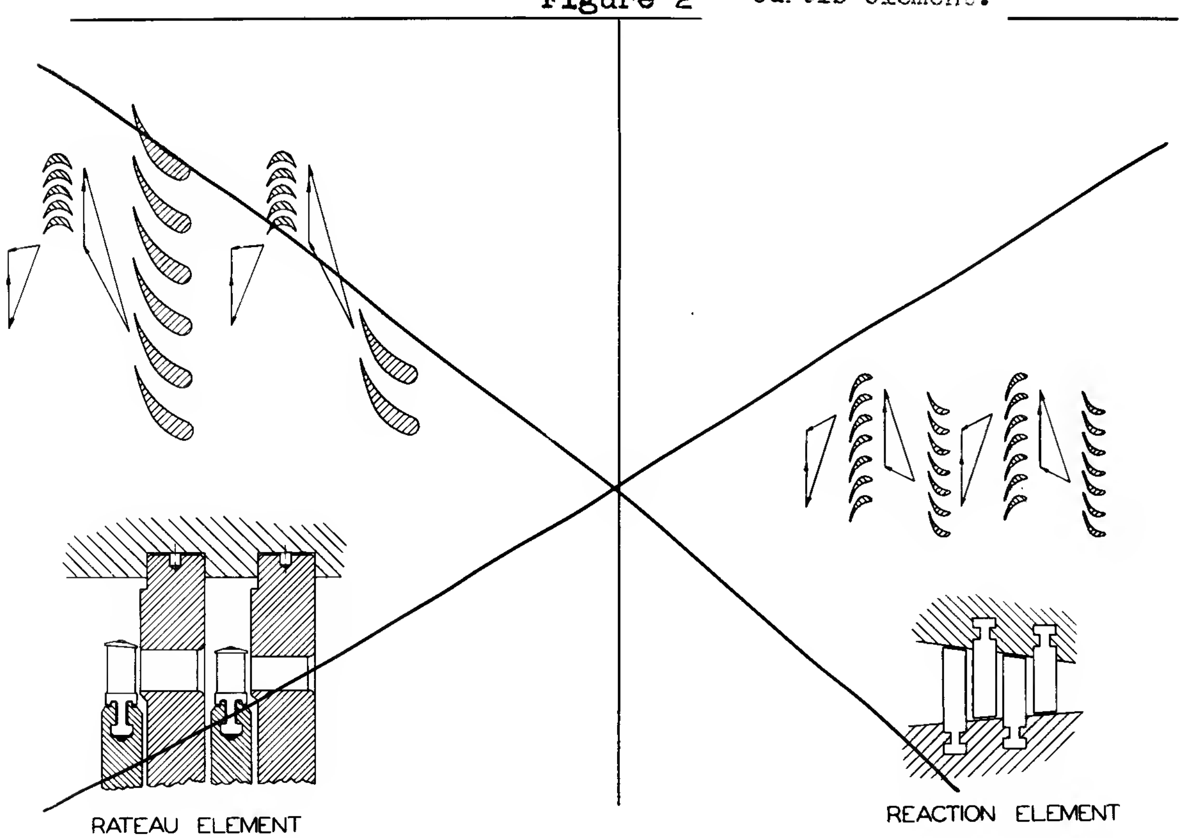


Figure 2

Failure of 1-1/2" wide Curtis blades.

Figure 3

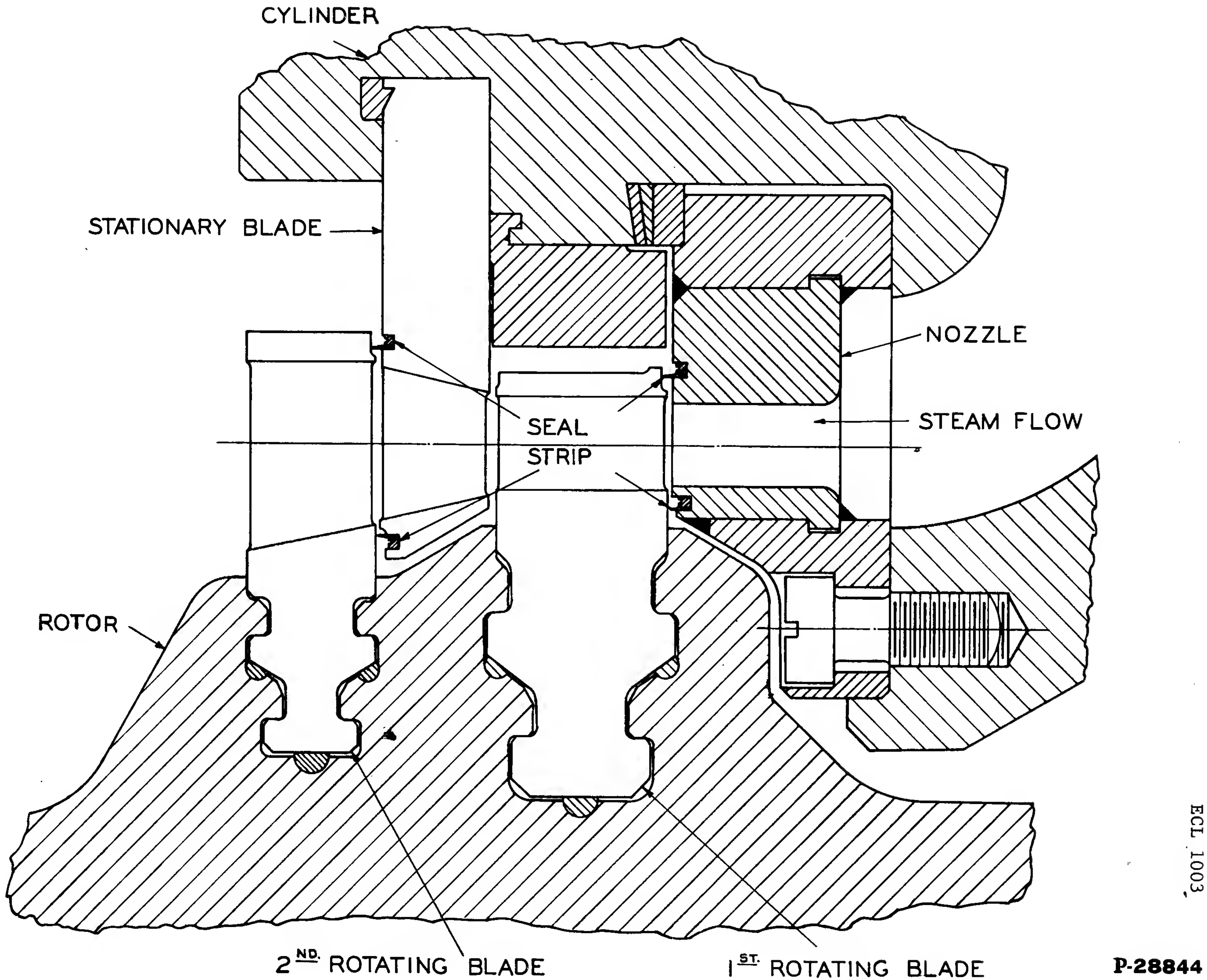


Figure 4 "Double T" impulse blades.

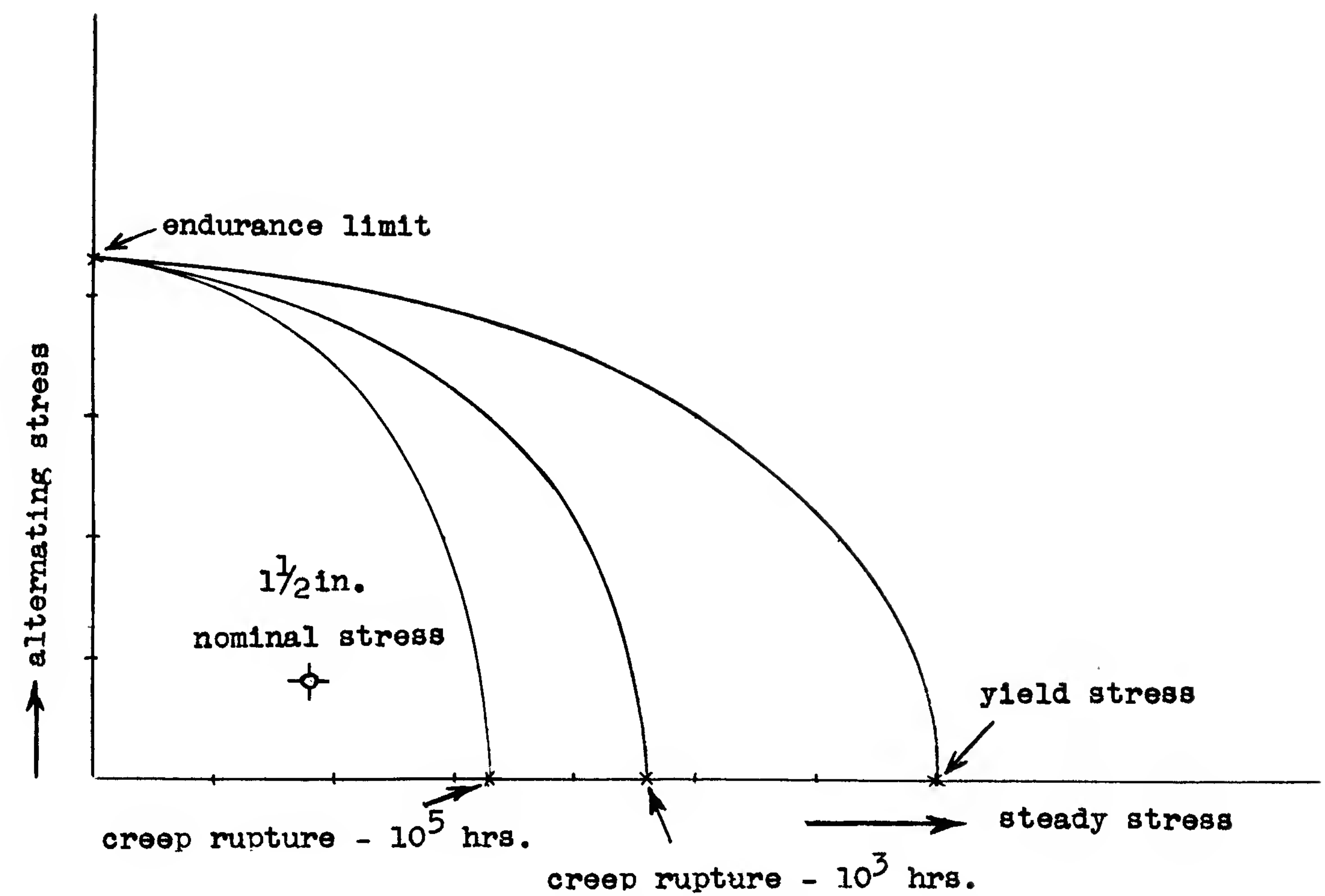


Figure 5a
Goodman diagram.

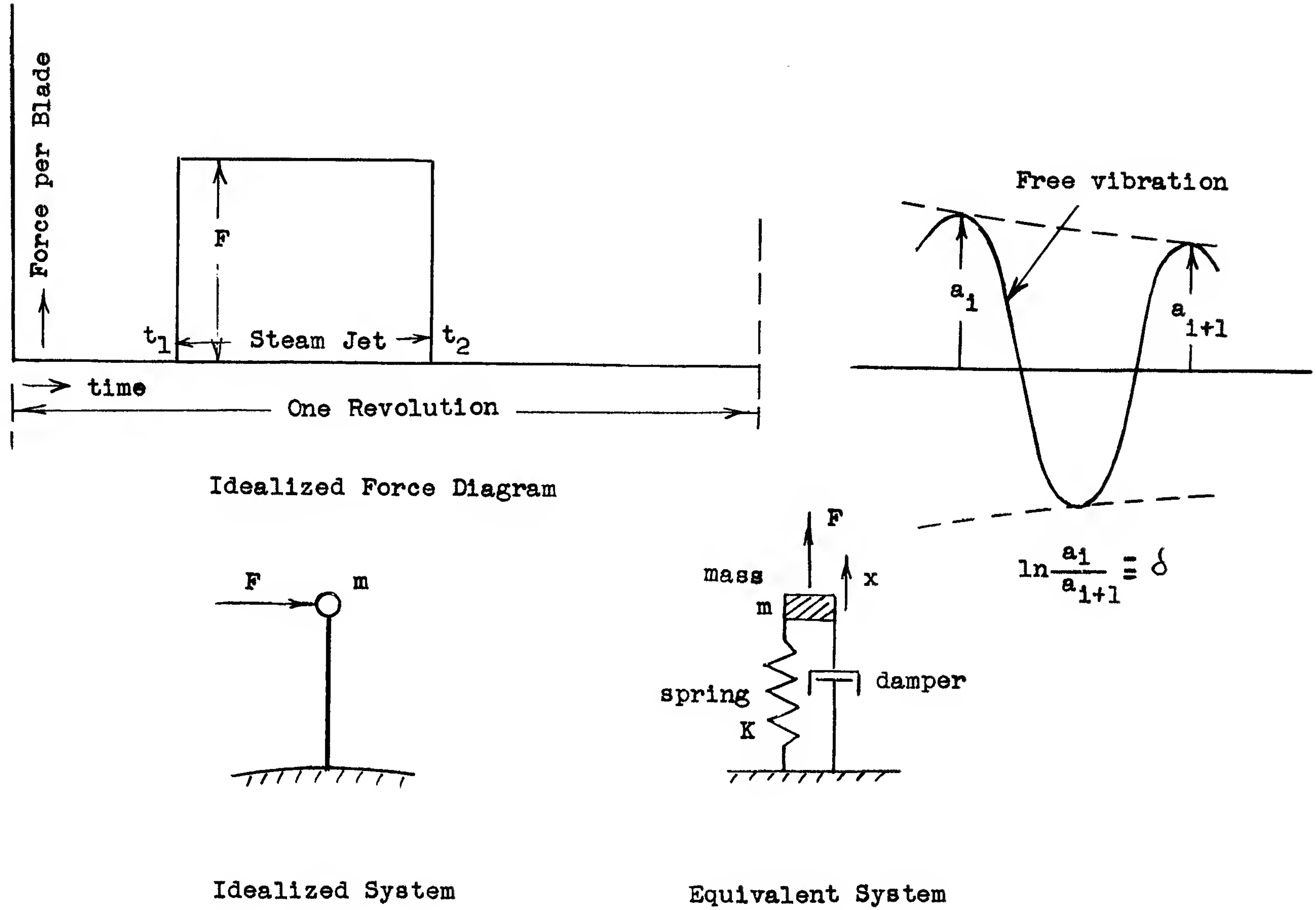
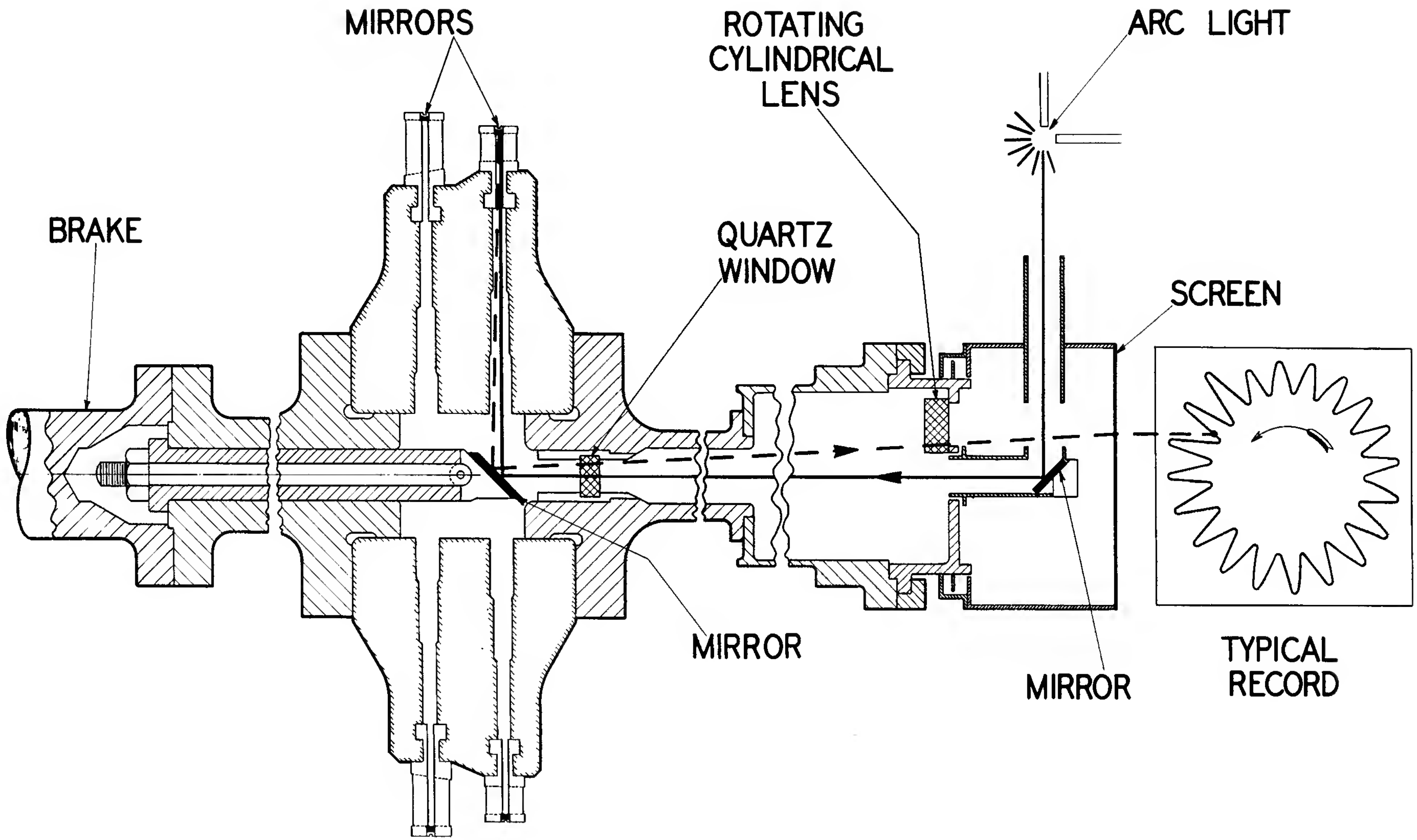


Figure 5 b

Idealized force diagram - idealized system.



OPTICAL SYSTEM
IMPULSE BLADE RESEARCH

Figure 6 Optical System

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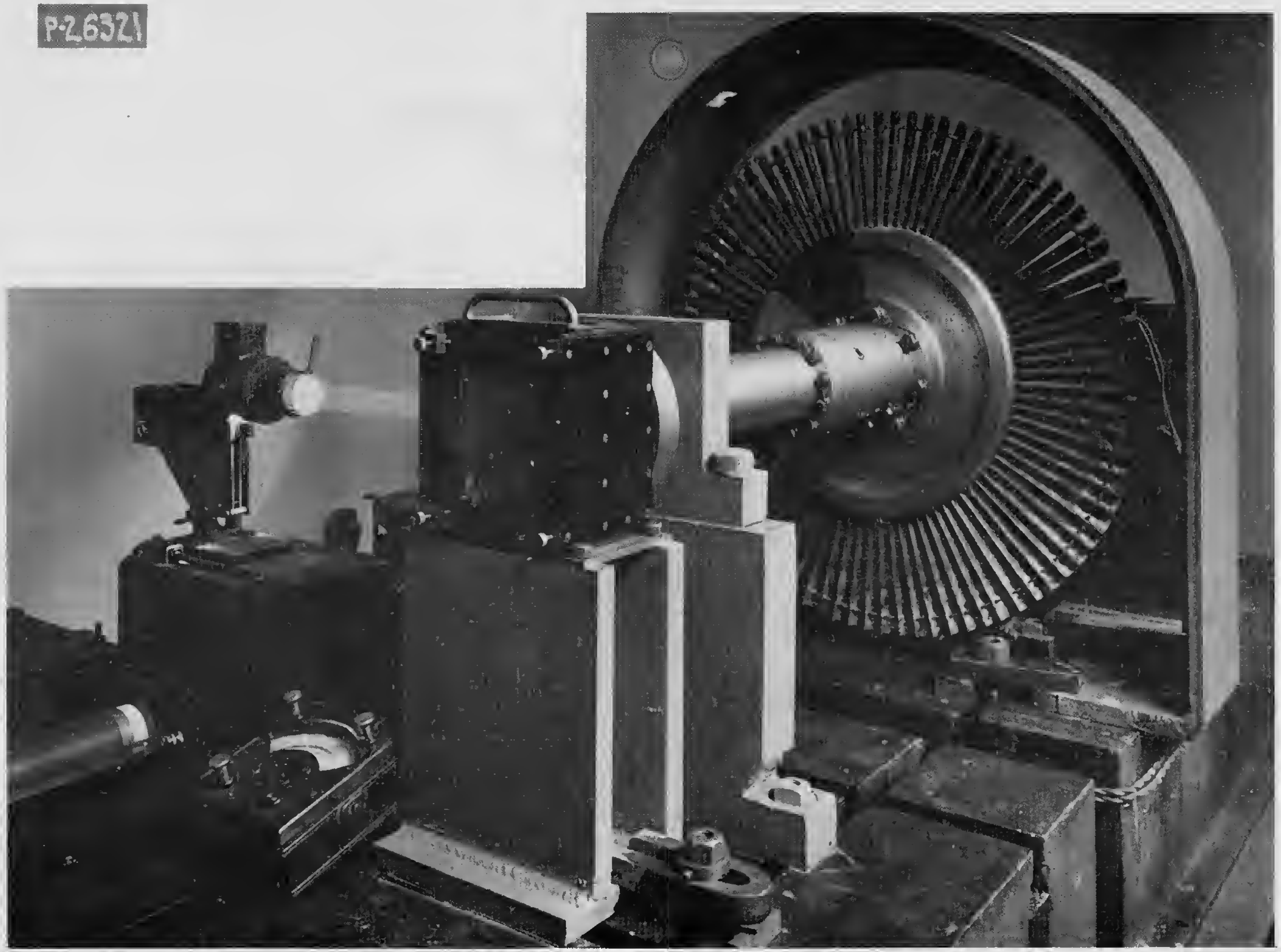

Experimental arrangement of Optical System.

Figure 7

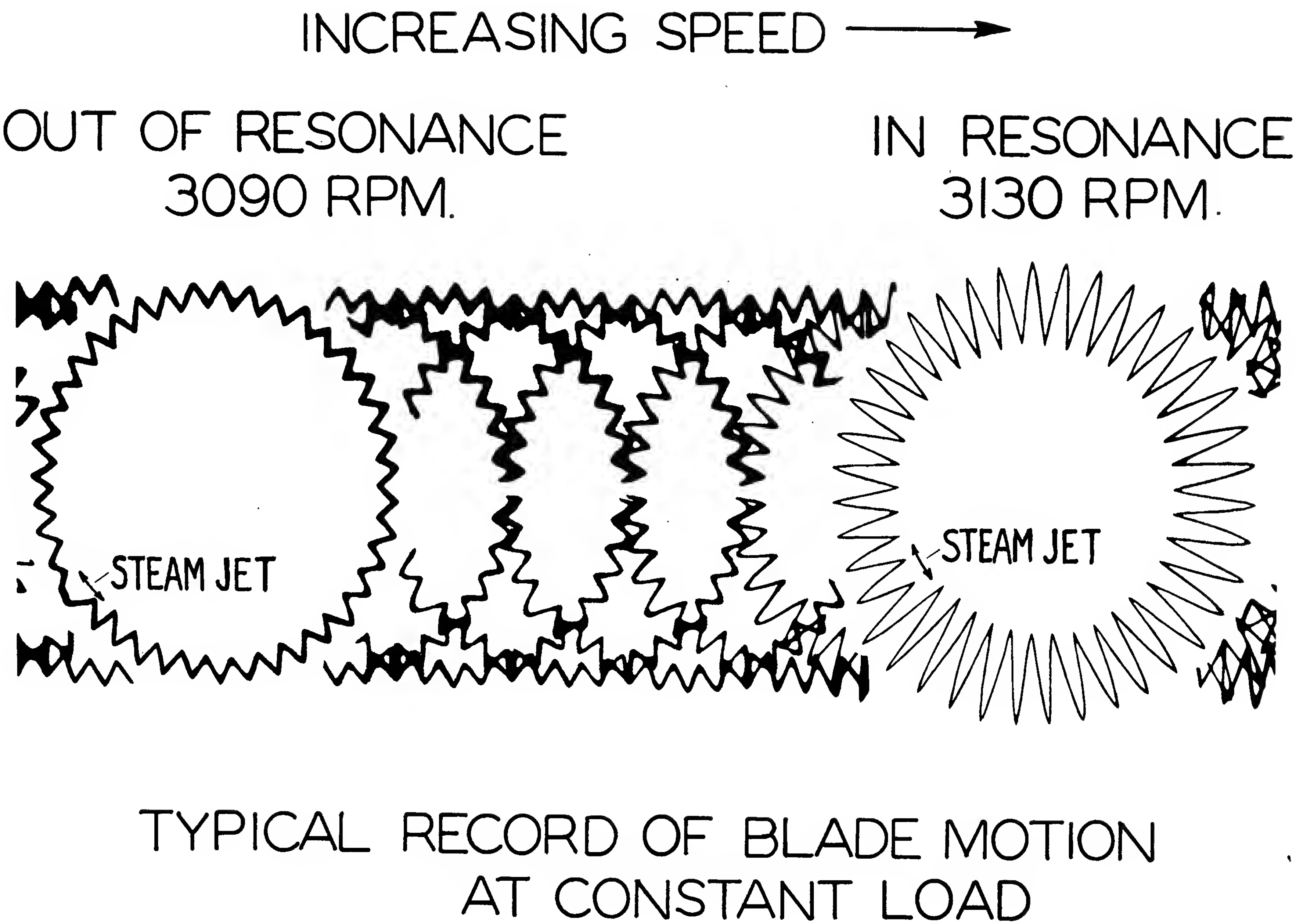

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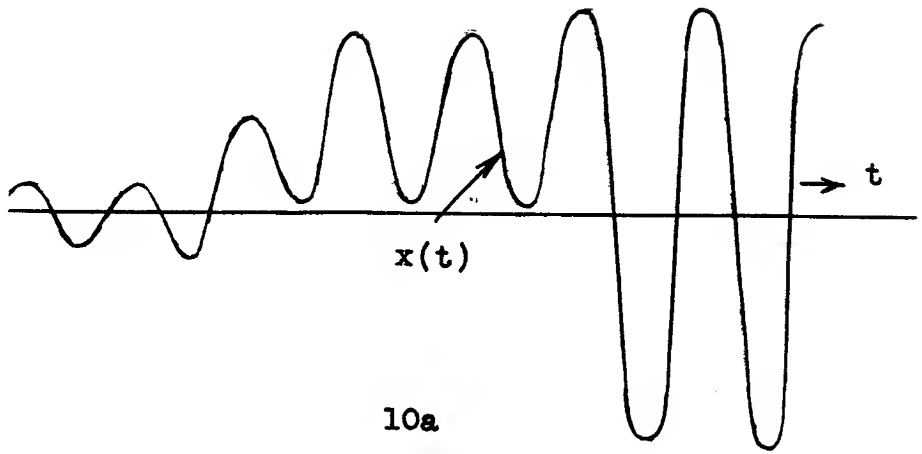
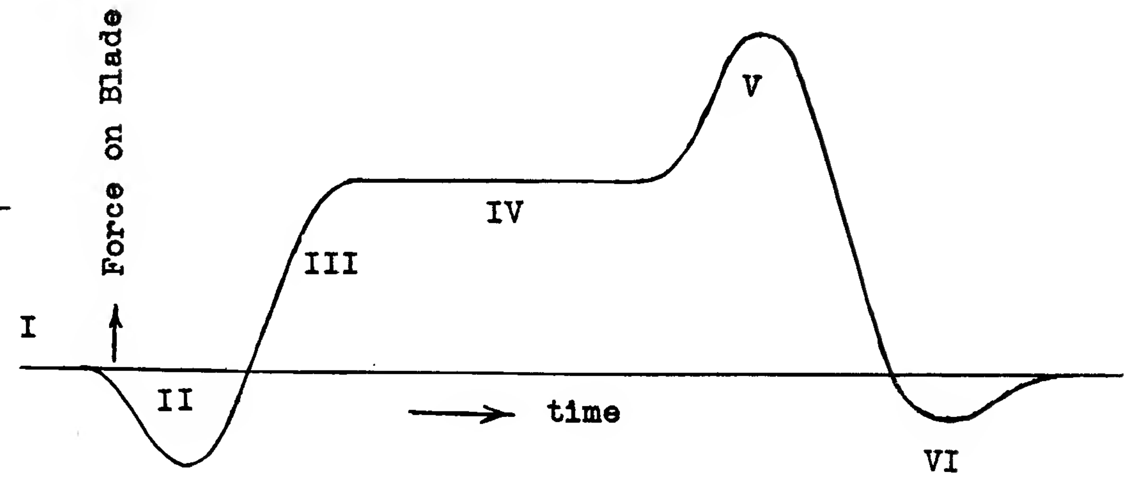
Figure 8 First experimental blade vibration record.

Figure 9

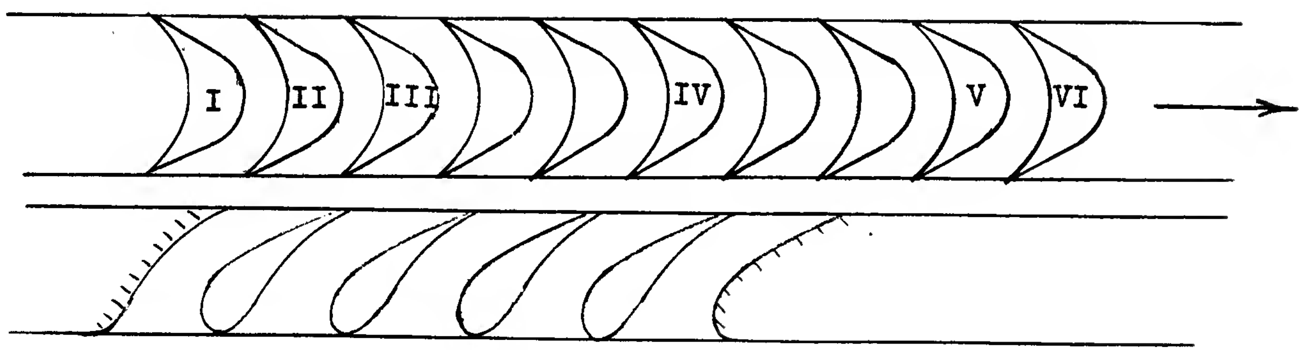
A mirror is being inserted at the tip of the blade in the full-scale experimental turbine. The young man in the picture is now Professor Donald F. Bradbury of the University of Rhode Island.



10a



10b



10c

Figure 10

Actual steam forçes.

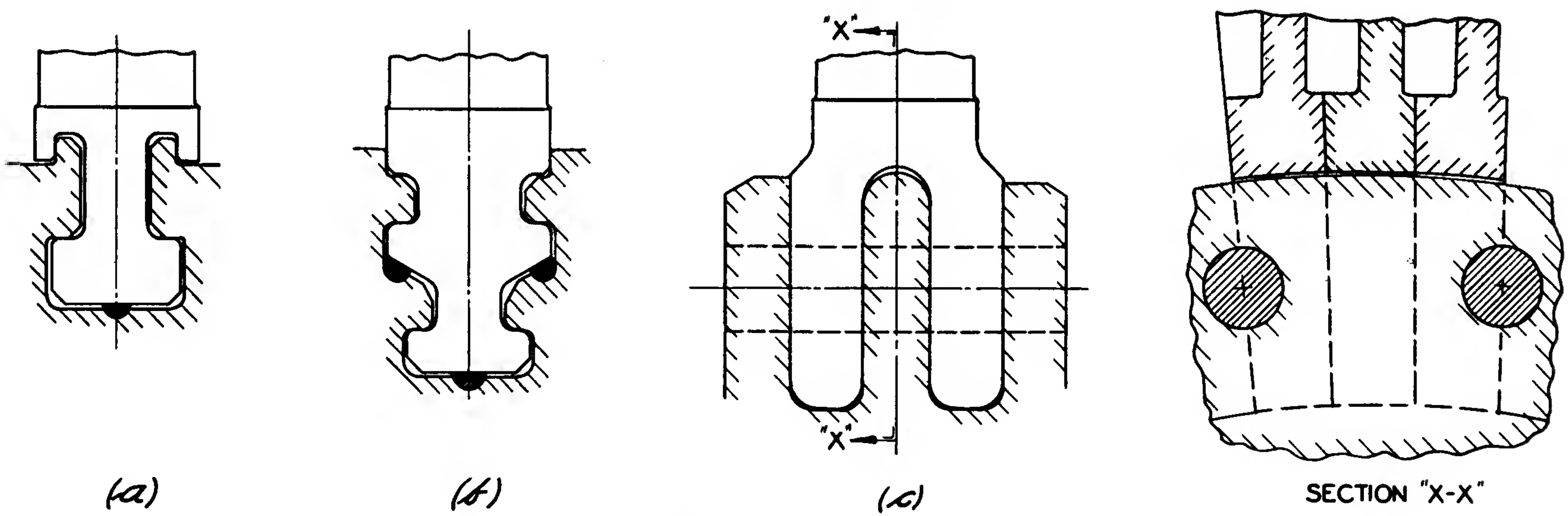


Figure 11 Impulse blade roots.

Figure 12

Side entry blade roots.

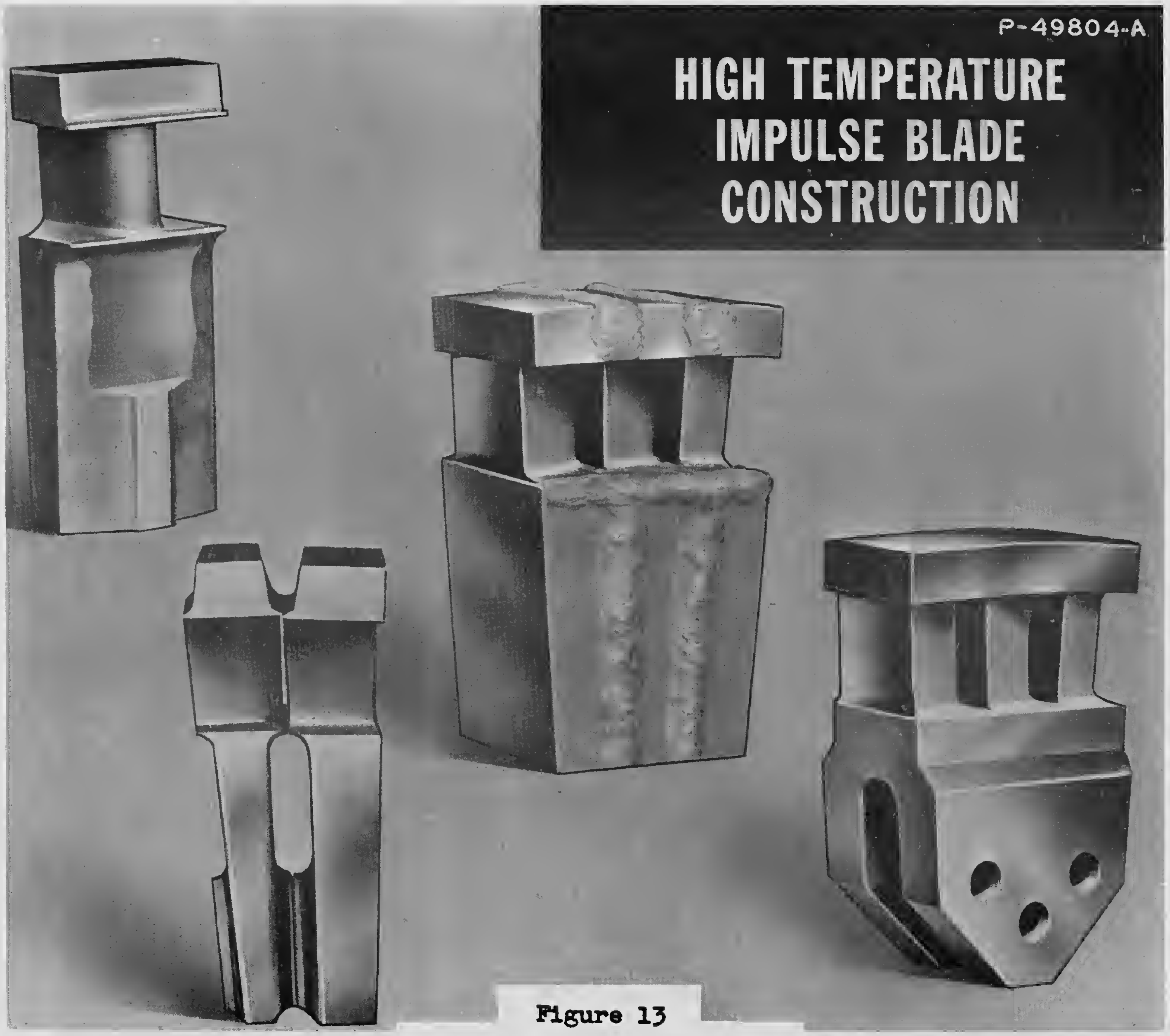


Figure 13

Multiple welded blade design.

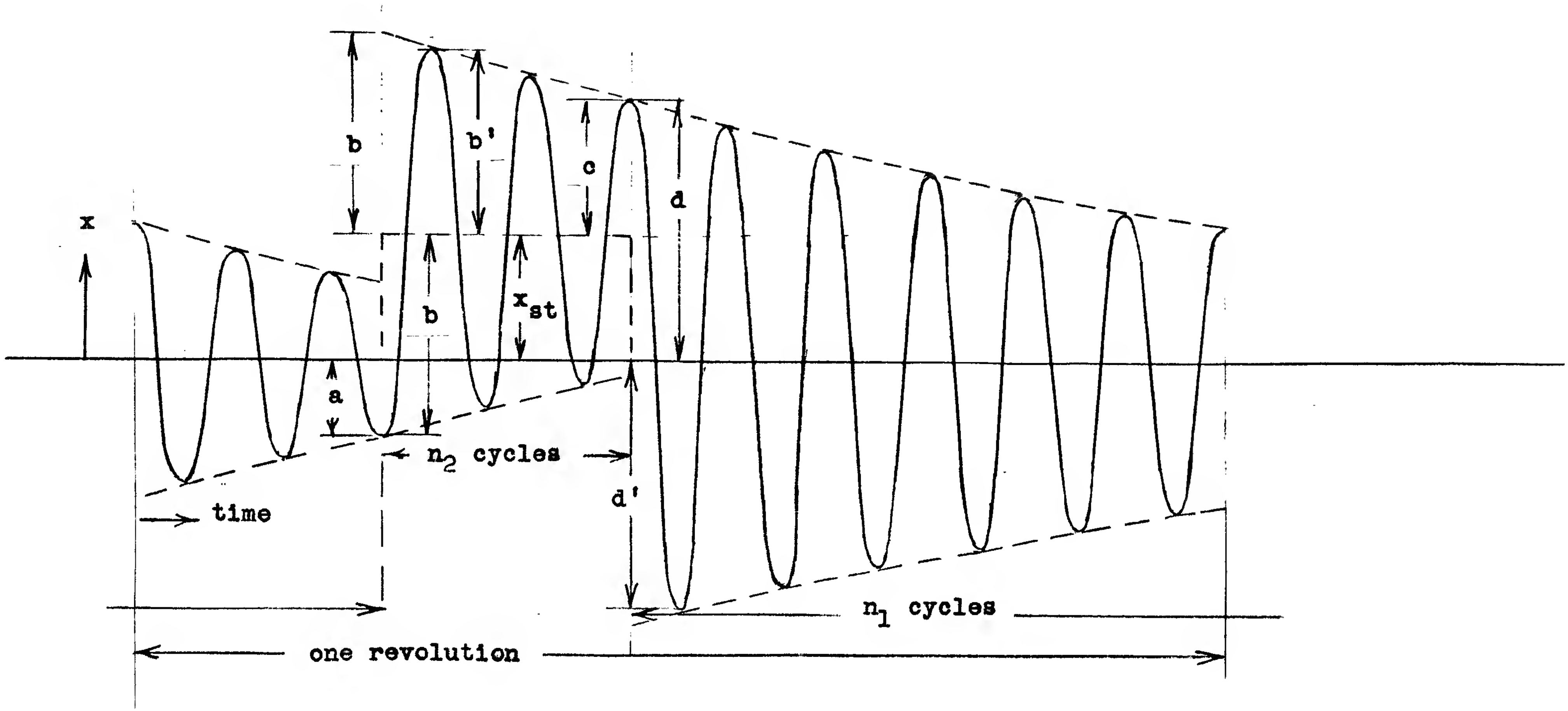


Figure 14
Blade vibration with idealized force diagram.

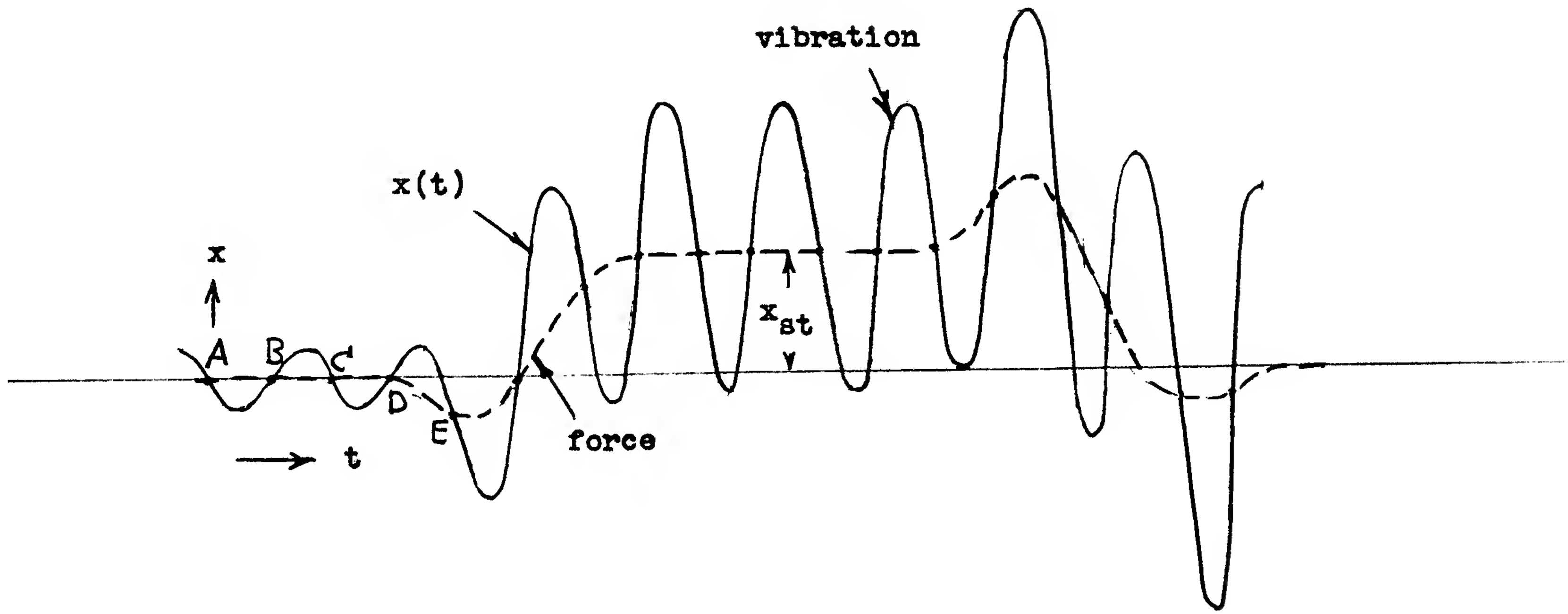


Figure 15

Constructing a steam force diagram.

INSTRUCTOR'S NOTE

for

CASE STUDY OF IMPULSE TURBINE BLADE FAILURE

This case study deals with the analysis and experimental investigation of turbine blade vibration due to intermittent steam loading. This problem has turned out to be very significant in steam turbine design.

The vibration analysis presented can easily be mastered by senior or first year graduate students. Depending on the instructor's approach, the case study can be used in several ways:

(a) As corollary reading material in a course in vibrations or in machine design. The material presented is self-sufficient for this purpose.

(b) As a means to challenge the student to work out various analytical and experimental problems by himself, the solutions to be discussed by the instructor. The instructor may want to pose additional, more sophisticated questions suggested by the nature of the problem. The history presented deals with some aspects of failure diagrams and with fatigue failures due to combined stress, and the instructor may want to elaborate on these subjects. Several questions or assignments have been included, and solutions are provided in appendices. Other questions are left open-ended, giving the instructor the opportunity to to test the student's ability to tackle design problems.

The instructor may want to take advantage of the case presented to point out that

(1) engineering history proves that many problems are not recognized until someone stumbles onto them.

(2) there is quite a difference between the typical classroom problem with the one right answer, and the many-faceted problems encountered in design.